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DAMPING OF VIBRATIONS IN CYLINDRICAL SHELLS OF GLASS AND PLASTIC

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DAMPING OF VIBRATIONS IN CYLINDRICAL SHELLS OF GLASS AND PLASTIC

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ABSTRACT. External losses were held to a minimum by setting the amplitudes of the radial displacements of a glass and plastic cylinder at resonant frequencies. The damping coefficient of such a shell exceeds analogous values for a steel shell by a factor of approximately 15.

The study of the damping of vibrations in shells is hindered by the complexity of the compressed--deformed state of the shells and the lack of simple relationships between the parts themselves and the corresponding wave numbers. /121*

Contradictory statements are found in [1-3] regarding damping of vibrations in cylindrical shells, and the nature of the change in the damping coefficient as a function of wave numbers. Experiments on aluminum shells [2] indicated that damping coefficient values are constant over a wide range of frequencies. Similar studies of the damping of vibrations in steel shells [1] revealed the existence of a relationship between the damping coefficient and the wave numbers. It is important to note, however, that these experimental studies were conducted at normal pressure and density of the ambient medium, without considering the amplitudes of the linear velocities. Hence, it was not possible to divide the energy losses incurred through vibration into internal and external ones, caused by resistance of the medium and acoustic radiation. /122

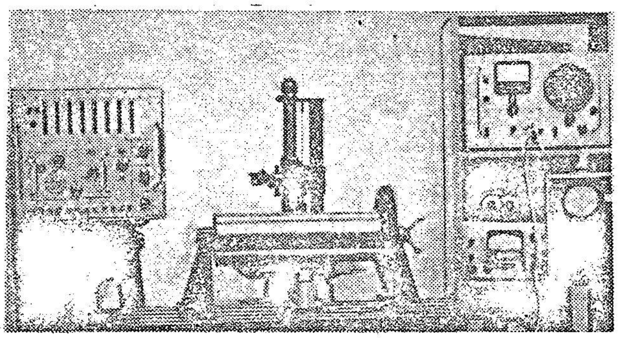

Figure 1.

In this report, the external losses did not vary during the investigation. This was accomplished by setting the amplitudes of the radial displacements at resonant frequencies so that the amplitudes of the linear

*Numbers in margin indicate pagination in the foreign text.

velocities assumed values that remained close together over the entire range of investigated frequencies.

The study of the processes of damping of vibrations in freely open shells of glass and plastic was conducted on an experimental setup of which an overall view of given in Figure 1. The shell is mounted on end supports at the centers of an adjustable mechanism. The vibrations are excited in the shell by one or (if necessary) several electromagnets, powered by a GZ-12 audio-frequency generator. Steel disks are attached to the shell at the points where harmonic force is applied.

The tests were conducted using a three-layered shell (496 mm long, 163 mm i.d., 1 mm wall), produced by winding No. 9,6 alkali-free glass thread on a special mandrel. The direction of the threads in the inner and outer layers coincides with the length of the shell, and in the middle layer with the generatrix of the shell. The binder used to make the shell contained ED-6 epoxy resin, Type "A" bakelite varnish, and BF-4 cement. The shell was cured at 423°K for 30 hours.

The logarithmic decrement of the vibrations as determined from oscillograms of the free damping vibrations of the shell were used as the damping characteristic. The oscillograms were recorded as follows: the shell was first brought to the resonance and the position of one of the antinodes of the /123
vibrations was determined, to which a miniature piezoelectric accelerometer weighing 3.2 g was then attached. The signal from this sensor passed through a U2-6 amplifier to the corresponding loop of an MPO-2 oscillograph. The shell was again brought to resonance, the recording was made, and the power circuit of the vibrator was disconnected. In this way, the resonance mode and the subsequent damping oscillatory process were recorded. Figure 2 shows typical vibrograms of free damping vibrations for f_{1-4} = 429 Hz; f_{1-5} = 668 Hz, and f_{3-5} = 976 Hz.

The amplitude values of the changes in the antinodes of the vibrations in the resonance mode are calculated by the formula

$$w = \frac{248.5}{f^2_{m-n}} K,$$

where ω is the amplitude of the vibrations in mm, K is the acceleration in cm/sec^2; f_{m-n} is the frequency of the vibrations in Hz.

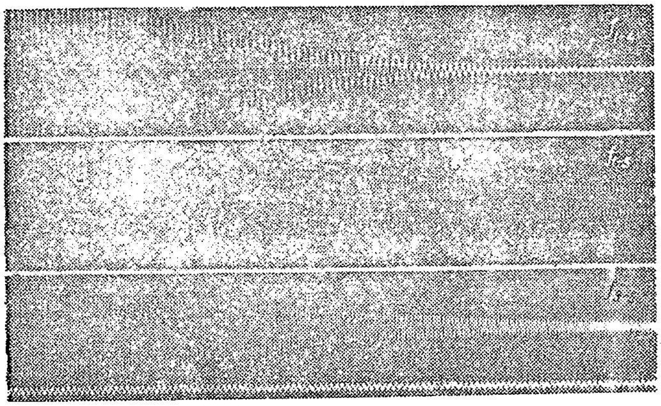

Figure 2.

The magnitude of the acceleration is determined by a PIU-1M piezoelectric accelerometer and a Type PDU-1 sensor, having an amplitude greater than 15 μ, with the aid of a cathetometer with an accuracy of ±1 μ.

The results of an analysis of the oscillograms of the damping vibrations, obtained for amplitudes of displacements in resonance within the limits of 5 ÷ 20 μ, are shown in Figure 3 in the form of curves, representing the logarithmic decrement as a function of the wave numbers m (the number of half-waves of deformations in the direction of the generatorix of the shell) and n (the number of waves of deformations along the length of the shell). These curves have a complex nature and have two extrema. The maximum values of the logarithmic decrements of the oscillations are observed at wave numbers corresponding to the minima of the frequency curves. /124
The largest value for all the decrements obtained is related to the minimum (fundamental) frequency of the vibrations of the shell. The minima of the curves in Figure 3 are much more weakly pronounced and correspond to mich higher values of n.

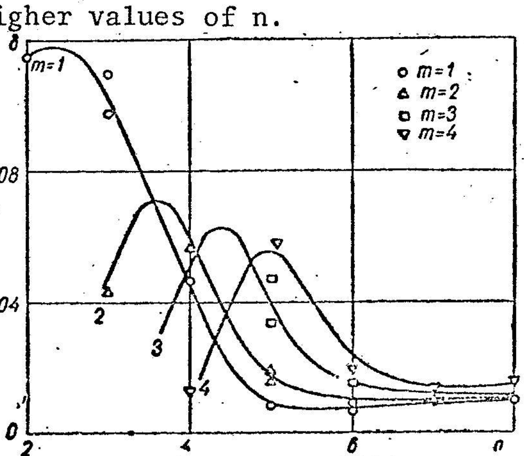

Figure 3.

Figure 4 shows the curves representing the damping coefficient ($h = \delta f_{m-n}$) as a function of the wave numbers m and n. The curves have their minima at the same values of n as the curves of the logarithmic decrements in Figure 3; the nature of these curves is in good agreement qualitatively with the curves in [1].

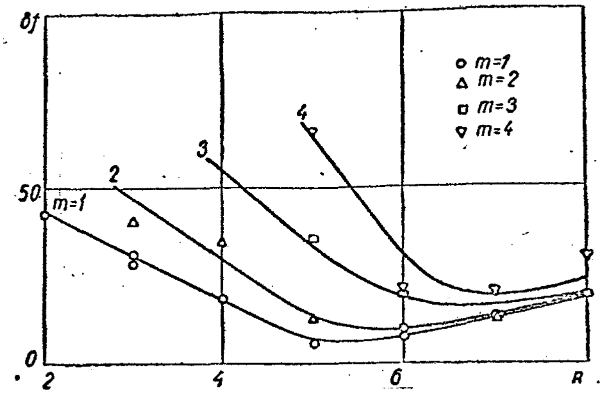

Figure 4.

In the quantitative relationship the damping coefficient of the shell investigated in this paper exceeds the analogous values for a steel shell by a factor of approximately 15.

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